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A review of flow and heat transfer characteristics in curved tubes

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Abstract

The performance of heat exchangers can be improved to perform a certain heat-transfer duty by heat transfer enhancement techniques. In general, these techniques can be divided into two groups: active and passive techniques. The active techniques require external forces, e.g. electric field, acoustic or surface vibration, etc. The passive techniques require fluid additives or special surface geometries. Curved tubes have been used as one of the passive heat transfer enhancement techniques and are the most widely used tubes in several heat transfer applications. This article provides a literature review on heat transfer and flow characteristics of single-phase and two-phase flow in curved tubes. Three main categories of curved tubes; helically coiled tubes, spirally coiled tubes, and other coiled tubes, are described. A review of published relevant correlations of single-phase heat transfer coefficients and single-phase, two-phase friction factors are presented.

Keywords: Curved tube; Heat transfer characteristics; Flow characteristics; Helically coiled tube; Spirally coiled tube; Pressure drop; Heat transfer coefficient; Friction factor

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1. Introduction

Heat exchangers are devices that are commonly used to transfer heat between two or more fluids of different temperatures. They are used in a wide variety of applications, e.g. refrigeration and air-conditioning systems, power engineering and other thermal processing plants.

Besides the performance of the heat exchanger being improved, the heat transfer enhancement enables the size of the heat exchanger to be considerably decreased. In general, the enhancement techniques can be divided into two groups: active and passive techniques. The active techniques require external forces, e.g. electric field, acoustic, surface vibration. The passive techniques require special surface geometries or fluid additives. Both techniques have been used for improving heat transfer in heat exchangers. Due to their compact structure and high heat transfer coefficient, curved tubes have been introduced as one of the passive heat transfer enhancement techniques and are widely used in various industrial applications. Helical and spiral coils are well known types of curved tubes which have been used in a wide variety of applications, for example, heat recovery processes, air conditioning and refrigeration systems, chemical reactors, food

Nomenclature

D	diameter of coil (m)
d	diameter of tube (m)
De	Dean number
D_h	hydraulic diameter (m)
d_i	inside diameter of tube (m)
d_o	outside diameter of tube (m)
$d_{i,out}$	inner diameter of outer tube (m)
$d_{o,in}$	outer diameter of inner tube (m)
f_c	friction factor in curved tube
f_s	friction factor in straight tube
Fr	Froude number
G	mass flux (kg/(m^2 s))
g	gravitational acceleration (m/s^2)
Gr	Grashof number
H	height of channel (m)
l	length of channel (m)
n	frequency of oscillation (s^{-1})
Nu	Nusselt number
p	pitch of coil (m)
Pr	Prandtl number
ΔP	pressure drop (Pa)
q	heat flux
R	radius of coil (m)
r	radius of pipe (m)
Re	Reynolds number
Re_{cri}	critical Reynolds number
Re_D	Reynolds number based on coil diameter
Re^*	modified Reynolds number
Sh	Sherwood number
Sc	Schmidt number
U_c	mean axial flow velocity (m/s)
U_G	superficial velocity of gas (m/s)
U_L	superficial velocity of liquid (m/s)
Vis	viscosity ratio, μ/μ_w
$\bar{W}_o$	oscillatory frequency number
w	width of the channel (m)
X	Lockhart–Martinelli parameter
x	vapor quality

Greek symbol

μ	viscosity (kg/(m s))
μ_l	viscosity of liquid (kg/(m s))

μ_w	viscosity at the surface (kg/(m s))
ρ	density (kg/m^3)
ρ_l	density of liquid (kg/m^3)
ρ_v	density of vapor (kg/m^3)
λ	aspect ratio, R/r
Δ	roughness of pipe
ϕ_L	pressure drop multiplier
β	helix angle
δ	curvature ratio
ν	kinematic viscosity (m^2/s)

and dairy processes. Although there are a large number of papers on the heat transfer and flow characteristics of curved tubes in the open literature, the review of these researches has not been performed.

The objective of this paper is to present a review of the work done on the characteristics of single-phase, two-phase heat transfer and flow in curved tubes. The curved tubes are classified under three main categories: helically coiled tubes, spirally coiled tubes, and other coiled tubes. A summary of published correlations of single-phase heat transfer coefficients and single-phase, two-phase friction factors are collected. This review can be indicated by the status of the research in this area which is important for the research in the future.

2. Helically coiled tubes

2.1. Heat transfer characteristics

2.1.1. Single-phase flow

Due to the curvature of the tubes, as fluid flows through curved tubes, centrifugal force is generated. A secondary flow induced by the centrifugal force has significant ability to enhance the heat transfer rate. Single-phase heat transfer characteristics in the helically coiled tubes have been widely studied by researchers both experimentally and theoretically. There are several advantages of the numerical method, e.g. large volume of the results obtained from the parametric studies, low cost. In addition, due to some complexity of the heat transfer processes in the helically coiled tubes, experimental studies are very difficult to handle. Numerical investigations are needed. Dravid et al. [1] numerically investigated the effect of secondary flow on laminar flow heat transfer in helically coiled tubes both in the fully developed region and in the thermal entrance region. The results obtained from predictions were validated with those obtained from experiments in the range in which they overlapped. A correlation for the asymptotic Nusselt numbers, Nu, was proposed as follows:

$$Nu = \left(0.65\sqrt{De} + 0.76\right)Pr^{0.175} \quad (1)$$

where Nu is the Nusselt number, De is the Dean number (50–200), and Pr is the Prandtl number (5–175).

Patankar et al. [2] discussed the effect of the Dean number on friction factor and heat transfer in the developing and fully developed regions of helically coiled pipes. Good agreements were obtained from comparisons between the developing and fully developed velocity profiles, the wall temperature for the case of axially uniform heat flux with an isothermal periphery obtained from calculation and those obtained from experiments. In the model mentioned above, the effects of the torsion and the Prandtl number were not taken into account. Yang et al. [3] presented a numerical model to study the fully developed laminar convective heat transfer in a helicoidal pipe having a finite pitch. The effects of the Dean number, torsion, and the Prandtl number on the laminar convective heat transfer were discussed. The helicoidal pipe was assumed to have uniform axial wall heat flux with a uniform peripheral wall temperature. The studied laminar flow of the incompressible Newtonian fluid was subjected to be hydrodynamically and thermally fully developed. The results revealed that the temperature gradient increased on one side of the pipe wall and decreased on the other side with increasing torsion. In the case of a fluid with a large Prandtl number, the Nusselt number was significantly decreased as torsion increased, but in the case of a fluid with a small Prandtl number, the Nusselt number declined slightly as the torsion increased.

Rabin and Korin [4] developed a new simplified mathematical model for thermal analysis of a helical heat exchanger for long-term ground thermal energy storage in soil for use in arid zones. The results obtained by solving a finite difference method were validated by experimental data. The various parametric studies such as thermal properties of the soil, cycle period, and height and pitch of the helical coil heat exchanger were studied. Zheng et al. [5] applied a control-volume finite difference method having second-order accuracy to solve the three-dimensional governing equations. The laminar forced convection and thermal radiation in a participating medium inside a helical pipe were analyzed. By comparing the numerical including and not including thermal radiation, the effects of thermal radiation on the convective heat transfer were investigated. They found that the thermal radiation could enhance the total heat transfer rate. Acharya et al. [6] numerically studied the phenomenon of steady heat transfer enhancement in coiled-tube heat exchangers due to chaotic particle paths in steady, laminar flow with two different mixings. The velocity vectors and temperatures fields were discussed. On the basis of the simulation data, a series of correlations of the spatially varying local and constant bulk Nusselts number were presented. Chen and Zhang [7] studied the combined effects of rotation (coriolis force), curvature (centrifugal force), and heating/cooling (centrifugal-type buoyancy force) on the flow pattern, friction factor, temperature distribution, and Nusselt number.

Rennie and Raghavan [8] simulated the heat transfer characteristics in a two-turn tube-in-tube helical coil heat exchanger. Various tube-to-tube ratios and Dean numbers for laminar flow in both annulus and in-tube were examined. The temperature profiles were predicted using a computational fluid dynamics package PHEONICS 3.3. The results showed that the flow in the inner tube at the high tube-to-tube ratios was the limiting factor for the overall heat transfer coefficient. This dependency was reduced at the smaller tube-to-tube ratio, where the influence of the annulus flow was increased. In all cases, as other

parameters were kept constant, increasing whether the tube Dean numbers or annulus Dean numbers resulted in an increase in the overall heat transfer coefficient.

Tarbell and Samuels [9] studied heat transfer characteristics in a helically coiled tube via the numerical solution of momentum and energy equations using the alternative direction-implicit technique. The results of the predicted asymptotic Nusselt number were compared with the experimental data of Dravid et al. and with Kalb and Seader's numerical results. The results showed that for $Pr < 5$, good agreement with Kalb and Seader's work was obtained. Later, Bolinder and Sunden [10] solved the paraborized Navier–Stokes and energy equations by using a finite-volume method. The steady fully developed laminar forced convective heat transfer in helical square ducts for various Dean and Prandtl numbers were analyzed. The mean Nusselt number and the local peripheral variation of the Nusselt numbers were presented for Prandtl numbers ranging between 0.005 and 500. In addition, correlations for the Nusselt number were proposed.

Sillekens et al. [11] employed the finite difference discretization to solve the paraborized Navier–Stokes and energy equations in a helically coiled heat exchanger. The effect of buoyancy forces on heat transfer and secondary flow was considered. It showed that, for the helically coiled tube with constant wall temperature boundary conditions, secondary flow induced by centrifugal and buoyancy forces affected the heat transfer rate. In their second paper, Rindt et al. [12] studied the development of mixed convective flow with an axial varying wall temperature. The results were compared with the constant wall temperature boundary condition. It was found that for all Grashof numbers, both heat transfer and secondary flow exhibited a wavy behaviour in axial direction. For higher Grashof numbers, for the case with an axial varying wall temperature, this phenomenon diminished due to stabilizing stratification effects. Recently, Lemenand and Peerhossaini [13] simplified the Navier–Stokes and energy equations as a thermal model to predict heat transfer rates in a twisted pipe of two tube configurations, helically coiled and chaotic. Based on the large database obtained from a numerical model, a correlation of the Nusselt number for coil geometry with an alternating plane of curvature was obtained.

As mentioned above, the heat transfer characteristics in helically coiled tubes have been done by a number of researchers. Among these studies, Yang and Ebadian [14] solved the $k-\varepsilon$ model to analyze the fully developed turbulent convective heat transfer in a circular cross-section helicoidal pipe with finite pitch. The results showed that as the pitch of the coil increased, the temperature distribution in the cross-section was asymmetrical. In the case of laminar flow, an increase in the Prandtl number would diminish the effect of torsion on the heat transfer. In addition, it was found that the pitch effect would be augmented as the flow rate increased. In a similar work, Lin and Ebadian [15] applied the standard $k-\varepsilon$ model to investigate three-dimensional turbulent developing convective heat transfer in helical pipes with finite pitches. The effects of pitch, curvature ratio and Reynolds number on the developments of effective thermal conductivity and temperature fields, and local and average Nusselt numbers were discussed. The results obtained from the model were in good agreement with the existing experimental data. Their second paper [16], presented the effects of inlet turbulence level on the development of three-dimensional turbulent flow and heat transfer in the entrance region of a helically coiled pipe for constant wall temperature and uniform inlet conditions. The Control-Volume Finite Element Method with an unstructured non-uniform grid system was used to solve

the governing equations. The results showed that at distances far from the entrance, the inlet turbulence level did not affect the bulk turbulent kinetic energy. The influence of the turbulence level on the development of the friction factor and Nusselt number was significant only for a short axial distance from the entrance.

In fact, results obtained from the mathematical model must be validated by comparing with experimental data. There are various experimental studies concerning heat and mass transfer characteristics in helically coiled pipes. Garimella et al. [17] presented average heat transfer coefficients of laminar and transition flows for forced convection heat transfer in coiled annular ducts. Two different coil diameters and two annulus radius ratios of test sections were used in their experiment. They found that the heat transfer coefficients obtained from the coiled annular ducts were higher than those obtained from a straight annulus, especially in the laminar region. Figueiredo and Raimundo [18] experimentally investigated the thermal response of a hot-water store and the thermal discharge characteristics from heat exchanger coils placed inside. The classical cylindrical coil and the flat spiral coil were investigated. The results indicated that the efficiency of flat spiral coil was higher than that of a cylindrical one. The results from comparison between the model and experiments were in good agreement.

Inagaki et al. [19] carried out experiments to investigate the flow-induced vibration, heat transfer and pressure drop of helically coiled tubes of an intermediate heat exchanger for a high-temperature engineering test reactor. Air was used as a working fluid. The heat exchanger model consisted of 54 helically coiled tubes separated into three layers. The results showed that the forced convective heat transfer of the tube outside was a function of $Re^{0.51}Pr^{0.3}$. The heat transfer rates between a helically coiled heat exchanger and a straight tube heat exchanger were compared by Prabhanjan et al. [20]. The experimental setup consisted of a helical coil of 10 turns with a 15.7 mm inner diameter, wall thickness of 1.2 mm, and no pitch. The helical diameter and the stretched length of the coil were 203 mm and 6.38 m, respectively. Results showed that the geometry of the heat exchanger and the temperature of the water bath surrounding the heat exchanger affected the heat transfer coefficient.

Besides that, experimental data was mostly used to validate the simulation; it was used to find important parameters such as heat and mass transfer coefficients. Developed correlations of the heat and mass transfer heat transfer coefficients and some experimental studies have been published in open literature. Each correlation was given with suitable validity ranges of the parameters. The free convection mass transfer characteristics of rings and helical coils were studied by Sedahmed et al. [21]. The following correlation was found to fit with mass transfer data for tubular rings within the range $5.5\times10^5 < Sc\cdot Gr < 9.4\times10^8$. The deviations from the single ring data of mass transfer data at the outer surface of helical coils depend on the number of turns per coil. The maximum deviation was found to be 12%.

$$Sh = 0.55(Sc\cdot Gr)^{0.25} \tag{2}$$

where Sh is the Sherwood number, Sc is the Schmidt number, and Gr is the Grashof number

Havas et al. [22] determined heat transfer coefficients of helical coils in agitated vessels. A modified Reynolds number was introduced into the heat transfer equation.

All 179 experimental data were regressed to obtain the correlation as follows:

$$Nu = 0.187 Re_{\mathrm{o}}^{0.688} Pr^{0.36} Vis^{0.11} \left(\frac{d_{\mathrm{a}}}{D_{\mathrm{v}}}\right)^{0.62} \tag{3}$$

For $1.3\times10^3 < Re < 1.6\times10^5$, $3.2\times10^2 < Re_o < 3.5\times10^4$, $2.7\times < Pr < 124$, $0.16 < Vis < 2.9$, $0.25 < d_a/D_v < 0.4$, $0.03 < d_o/D_v < 0.051$ where Re_{o} is the modified Reynolds number, $d_{\mathrm{o}} \cdot d_{\mathrm{a}} \cdot n/\rho$, Re is the Reynolds number,

$$\frac{d_a^2 \bullet n \bullet \rho}{\mu}$$

Vis is the viscosity ratio, μ/μ_{w}, d_{a} is the diameter of the agitator, d_{o} is the outside diameter of the tube, n is the agitator rotational frequency, D_{v} is the diameter of the agitated vessel.

The use of both active and passive techniques to enhance the heat transfer rate was reported by Cengiz et al. [23]. They studied the effect of rotation of helical pipes on the heat transfer rates and pressure drop for various air-flow rates. The coils were made from copper tubes with a diameter of 10 mm and a length of 3200 mm, respectively. The results showed that although the rotation caused an increase in pressure drop, the heat transfer rates were augmented. A correlation of the heat transfer coefficient for the case of rotating coils was proposed to represent the data within $\pm 10\%$ error. In their second paper [24], the heat transfer and pressure drop in a heat exchanger constructed by placing spring-shaped wire with varying pitch were studied. The results indicated that the Nusselt number increased with decreasing pitch/wire diameter ratio. On the basis of the experimental data for both empty helical pipes and helical pipes with springs installed inside, the correlations of the Nusselt number were presented as follows:

Empty helical pipes:

$$Nu = 0.0551 De^{0.864} Pr^{0.4} \tag{4}$$

for $1265 \le De \le 2850$, $Pr = 0.7$.

Helical pipes with spring:

$$Nu = 4.02 De^{0.785} Pr^{0.4} \left(\frac{H_{\mathrm{s}}}{d_{\mathrm{s}}}\right)^{-1.008} \tag{5}$$

for $1315 \le De \le 3200$, $Pr = 0.7$, where H_{s} is the spring pitch, and d_{s} is the spring diameter.

Xin and Ebadian [25] considered the effects of the Prandtl number and geometric parameters on the local and average convective heat transfer characteristics in helical pipes. Five helical pipes with different torsion and curvature ratios were tested with three different working fluids. The results showed that for the laminar flow region the peripheral Nusselt number changed significantly as the Prandtl and the Dean numbers increased. Based on the present data, new empirical correlations for the average fully developed were

obtained as follows:

$$Nu = \left(2.153 + 0.318De^{0.643}\right)Pr^{0.177} \quad \text{for } 20 < De < 2000,$$
$$0.7 < Pr < 175, \quad 0.0267 < d/D < 0.0884 \tag{6}$$

$$Nu = 0.00619Re^{0.92}Pr^{0.4}\left(1 + 3.455\frac{d}{D}\right) \quad \text{for } 5\times10^{3} < Re < 10^{5},$$
$$0.7 < Pr < 5, \quad 0.0267 < d/D < 0.0884 \tag{7}$$

Guo et al. [26] investigated the effects of pulsation upon transient convective heat transfer characteristics in a uniformly heated helical coiled tube for fully developed turbulent flow. The secondary flow mechanism and the effect of interaction between the flow oscillation and secondary flow were elucidated. A series of new correlations of the average and local heat transfer coefficients both under steady and oscillatory conditions were proposed in the following form:

For single-phase turbulent steady flow:

$$Nu = 0.328Re^{0.58}Pr^{0.4} \quad \text{for } 6000 < Re < 180,000 \tag{8}$$

For oscillatory single-phase turbulent flow:

$$Nu = 0.147\bar{W}_{\mathrm{o}}^{-0.31}Pr^{-4.4}\left(\frac{De}{1000}\right)^{0.82} \tag{9}$$

$$\bar{W}_{\mathrm{o}} = d_{\mathrm{i}}\bullet\sqrt{2n\pi/\mu_{1}} \quad \text{for } 0.003 < n < 0.05, \quad 25,000 < Re < 125,000, \tag{10}$$

where $\bar{W}_{\mathrm{o}}$ is the oscillatory frequency number, and n is the rotational frequency.

Only one work has considered the air-side heat transfer of helical pipes. Rahul et al. [27] determined the outside heat transfer coefficient from coiled tube surfaces in a cross-flow of air. The length of the test section was 1.5 m and the velocity of air ranged between 1 and 8 m/s. The influences of Reynolds number and pitch of the coiled tube surfaces were discussed. The results indicated that the pitch of the coil significantly affected the heat transfer coefficient. Based on the range of Reynolds numbers and pitch to tube diameter ratios used in their experiment, a correlation was developed as follows:

$$Nu_{\mathrm{o}} = 0.5186Re_{\mathrm{D}}^{0.595}\left[\frac{p}{d_{\mathrm{o}}}\right]^{0.857} \quad \text{for } 7000 < Re_{\mathrm{D}} < 55,000,\ 1.1275 < p/d_{\mathrm{o}}$$
$$< 1.8575. \tag{11}$$

2.1.2. Two-phase flow

Compared to the numerous investigations of the single-phase heat transfer, only a few works on the two-phase heat transfer characteristics in helically coiled tubes have been reported. Berthoud and Jayanti [28] studied the effects of pressure, coil diameter, mass flux

and heat flux on the dryout quality in helical coils using results gathered from various sources. The effects of these parameters on entrainment of the liquid, redeposition of the droplet, secondary flow, and phase change resulting from surface heat flux were also considered. Kang et al. [29] studied the condensation heat transfer and pressure drop characteristics of refrigerant HFC-134a flowing in a 12.7 mm helicoidal tube. Experiments were performed for the refrigerant mass fluxes from 100 to 400 kg/m^2/s, in the cooling water Reynolds number range of 1500–9000 at a fixed system temperature of 33 °C and the cooling tube wall temperature range of 12–22 °C. The effects of cooling wall temperature on heat transfer coefficients were also considered. However, with the increase of mass flux or the cooling water Reynolds number, the refrigerant-side heat transfer coefficients decreased. The following correlation of heat transfer coefficient was proposed based on the experimental data.

$$\frac{Nu}{Pr^{0.4}} = 2.3(Re^*)^{0.94} \tag{12}$$

$$Re^* = \frac{G \bullet x \bullet d_\mathrm{i}}{\mu_\mathrm{l}\sqrt{\rho_\mathrm{l}/\rho_\mathrm{v}}} \tag{13}$$

where G is the mass flux, x is the vapor quality, d_i the inside diameter of tube, Re^* is the modified Reynolds number, ρ_l is the liquid density, ρ_v is the vapor density, and μ_l is the liquid viscosity.

Recently, in their second paper, Yu et al. [30] investigated the effects of the different orientations of helical pipe on the condensation heat transfer of R-134a. The results revealed that the orientation of helical pipe has a significant effect on both refrigerant-side and overall heat transfer coefficients.

Guo et al. [31] investigated the effects of pulsation on transient convective heat transfer characteristics of steam–water two-phase flow in a helical-coil tube steam generator. The secondary flow and the effect of interaction between the flow oscillation and secondary flow were elucidated. The results showed that for pulsation flow, there exist considerable variations in the local and peripherally time-average Nusselt number. A correlation of the time average heat transfer coefficients under oscillatory flow conditions was proposed. Later, Yi et al. [32] studied the heat transfer characteristics and flow patterns under different filling ratios and heat fluxes of the evaporator section using small helically coiled pipes in a looped heat pipe. The glass and stainless pipe were used as an evaporator in the heat pipe. The results showed that the disturbance resulted in flow pulsation and the secondary flow augmented significantly the heat transfer rate and the critical heat flux. In addition, they proposed two correlations for predicting the heat transfer coefficient in the evaporator section before and after dryout occurs.

2.2. Flow characteristics

2.2.1. Single-phase flow

A secondary flow is induced due to the difference in the centrifugal force caused by fluid elements moving with different axial velocities [33]. The flow phenomenon in curved tubes is therefore more complex than in straight tubes. In addition, the pressure drop for

flow in curved tube is higher than that for in straight tube at the same flow rate and tube length. Many researches have been conducted regarding fluid flow in helically coiled tubes with circular cross-sections. Tarbell and Samuels [9] solved the equations of motion and energy to study flow characteristics in helical coils by using the alternating direction-implicit technique. The numerical results were compared with the experimental data of White [34], boundary layer analysis results of Mori and Nakayama [35], and numerical solution of Truesdell and Adler [36]. A correlation of friction factor representing the data within 3% was proposed:

$$\frac{f_c}{f_s} = 1.0 + \left[8.279 \times 10^{-4} + \frac{7.964 \times 10^{-3}}{\lambda}\right] Re - 2.096 \times 10^{-7} Re^2 \quad \text{for } 20 < De < 500,\ 3 < \lambda < 30 \tag{14}$$

where f_c is the friction factor for curved tube, f_s is the friction factor for straight tube, and λ is the ratio of the radius of curvature to radius of tube.

Numerous researchers have studied the effects of torsion and curvature of the tubes on the flow characteristics in curved tubes. Wang [37] proposed a non-orthogonal helical co-ordinate system to investigate the effects of curvature and torsion on the low-Reynolds number flow in a helical pipe. The results showed that when the Reynolds number was less than around 40, non-negligible effects were induced by curvature and torsion. However, when the Reynolds number was 1, a secondary flow consisting of a single recirculating cell was induced by the torsion while the curvature caused the increase of flow rate. These influences were completely different from the two recirculating cells and decreased flow rate at high Reynolds number. Huttl and Friedrich [38,39] applied the second order accurate finite volume method for solving the incompressible Navier–Stokes equations to study the effects of curvature and torsion on turbulent flow in helically coiled pipes. The incompressible Navier–stokes equations were expressed in an orthogonal helical coordinate system. The results showed that the flow quantities were affected by the pipe curvature. Although the torsion effect was less, it cannot be neglected. This is because it affected the secondary flow induced by pure curvature and resulted in an increase of fluctuating kinetic energy and dissipation rate.

Yamamoto et al. [40] studied the effects of torsions and curvatures on the flow characteristics in a helical tube. The experiments were carried out with three different dimensionless curvatures and seven different torsional parameters. The results showed that the torsions had a destabilizing effect on the flow. The critical Reynolds number at the onset of turbulence depends on torsional parameters. The results obtained from the experiments were compared with those obtained from the model of Yamamoto et al. [41]. In their third paper [42], they numerically studied the combined effects of rotation, torsion and curvature on the incompressible viscous steady flow through a helical pipe. The results showed that the rotation greatly affected the variation of the total heat flux when the direction of rotation was negative. In addition, in 2002 [43], they studied the secondary flow structure and stability of flow in a helical pipe with large torsion by using a numerical calculation of a fluid particle trajectory. The results obtained from the model were in good agreement with those of the experiments.

A practical friction diagram of helically coiled tube which accounts for the effect of diameter ratios was presented by Grundman [44]. The calculation was based on the equations of Mishra and Gupta [45,46]. Later, Hart et al. [47] presented a tube friction chart for laminar and turbulent flow for single-phase and two-phase flow through helically coiled tubes cover. The experiments were performed in a helically coiled glass tube with a 14.66 ± 0.04 mm tube diameter and 421 ± 2 mm coil diameters.

Visual observation of the flow pattern and Laser-Doppler Velocimetry measurements of laminar flow in a helical square duct with finite pitch were reported by Bolinder and Sunden [48]. The test section was milled from a solid PVC cylinder. The measured velocity profiles were in good agreement with profiles obtained from numerical calculations using the finite-volume method assuming a fully developed flow. In a similar work, Ujhidy et al. [33] used the laser technique for visualization of the laminar flow of water in coils and tubes containing twisted tapes and helical static elements. The secondary flow induced in the channel between the tube wall and the surface of a helical element was elucidated. Good agreement was obtained between the results from experiments and those from calculation.

Xin et al. [49] studied the effects of coil geometries and the flow rates of air and water on pressure drop in both annular vertical and horizontal helicoidal pipes. The test sections with three different diameters of inner and outer tubes were tested. The results showed that the transition from laminar to turbulent flow covers a wide Reynolds number range. On the basis of the experimental data, a correlation of the friction factor was developed.

$$f_c = 0.02985 + \frac{75.89\left[0.5 - \left(\tan^{-1}\left(\frac{De-39.88}{77.56}\right)\right)/\pi\right]}{\left(\frac{D}{d_{i,out}-d_{o,in}}\right)^{1.45}} \tag{15}$$

where $35 \leq De \leq 20{,}000$, $1.61 \leq d_{i,out}/d_{o,in} \leq 1.67$, $21 \leq D/(d_{i,out} - d_{o,in}) \leq 32$.

The maximum deviation of the friction factor from experiments and the correlation was found to be 15%.

In 2001, Ju et al. [50] used an HTR-10 steam generator to evaluate the hydraulic performance of small bending radius helical pipe. The results showed that the critical Reynolds number of helical pipe in a function of the Dean number was much greater than a straight pipe. All experimental data were regressed to obtain the friction factor correlations as follows:

For $De < 11.6$, it is laminar flow:

$$f_s = \frac{64}{Re}; \quad \frac{f_c}{f_s} = 1 \tag{16}$$

For $De > 11.6$, $Re < Re_{cri}$, it is laminar with large vortex:

$$f_s = \frac{64}{Re}; \quad \frac{f_c}{f_s} = 1 + 0.015 Re^{0.75}\left(\frac{d}{D}\right)^{0.4} \tag{17}$$

For $De > 11.6$, $Re > Re_{cri}$, it is turbulent flow:

$$f_s = \frac{0.316}{Re^{.25}} \quad \text{(smooth pipe)} \tag{18}$$

$$f_s = 0.1\left(1.46\frac{\Delta}{d} + \frac{100}{Re}\right)^{0.25} (\Delta: \text{ roughness of the pipe}) \tag{19}$$

$$\frac{f_c}{f_s} = 1 + 0.11Re^{0.23}\left(\frac{d}{D}\right)^{0.14} \tag{20}$$

Guo et al. [51] studied frictional pressure drops of single-phase water flow in two helically coiled tubes at four different helix axial inclinations angles. The results indicated that the helix axial angles have insignificant effect on the single-phase frictional pressure drop. All measured data were fitted to obtain a new friction factor correlation in the following form:

$$f_c = 2.552Re^{-0.15}\left(\frac{d}{D}\right)^{0.51} \tag{21}$$

A pressure drop correlation in terms of Euler number, Reynolds number, and geometrical group for steady isothermal flow of Newtonian fluids in helically coiled pipes was proposed by Ali [52]. The test sections with eight different geometrical parameters were built and tested. The results showed that the Reynolds number and geometrical number affected the fanning friction factor. Downing and Kojasoy [53] studied the effect of curvature on the pressure drop of R-134a flowing through miniature helical channels. Eight different curvatures and channel sizes of helical channel employed as test sections were examined to cover a wide-range of flow conditions. The measured data were compared with the Hart et al.'s correlation [47].

2.2.2. Two-phase flow

Compared to single-phase flow, two-phase flow characteristics and frictional pressure drop are more complex and important for engineering practice. A number of correlations for two-phase frictional pressure drop have been found in the literature. The studies of the two-phase flow in helically coiled tubes mostly use the correlations based on the Lockhart–Martinelli parameter. Kasturi and Stepanek [54] determined pressure drop and void fraction for the two-phase co-current flow of gas-liquid in a helical coil. Air–water, air–corn-sugar–water, air–glycerol–water, and air–butanol–water were used as working fluids. The measured data were compared with the calculation results from the Lockhart–Martinelli correlation, Duckler's correlation and Hughmark's correlation. Their second paper [55] proposed the correlations for void fraction and pressure drop in terms of new parameters. They proposed that the advantage of the proposed correlation was that it accounted more fundamentally for the complex behaviour of the two-phase flow than the simple correlation in terms of Lockhart–Martinelli parameters. Rangacharyulu and Davies [56] studied the pressure drop and holdup for co-current upwards flow of air-liquid in helical coils. Water, glycerol and isobutyle alcohol were used as working fluids. The flow rates of air and liquid were varied from 1 to 10 m^3/h, and 0.04 to 0.75 m^3/h, respectively. Based on the modified Lockhart–Martinelli parameter, a new correlation for the two-phase frictional pressure drop was presented.

Awwad et al. and Xin et al. [57,58] investigated the air–water two-phase flow in horizontal and vertical helicoidal pipes, respectively. Four different inside diameters of tubing and two different outside diameters of the cylindrical concrete forms were used for making the helicoidal pipe with various configurations. For horizontal helicoidal pipes [57], it was found that the superficial velocities of air or water had significant effect on the pressure drop multiplier, while the helix angle had insignificant effect and the pipe and coil diameters had a certain effect only at low flow rates. For vertical helicoidal pipes [58], at low flow rates in small aspect ratios, the Lockhart–Martinelli parameter and the flow rates affected the two-phase pressure drop. The void fraction was influenced by the geometric parameters affecting the frictional pressure drop. Based on their experimental data for both vertical and horizontal helicoidal pipes, correlations of the frictional pressure drop multiplier for two-phase flow were proposed as follow:

For horizontal helicoidal pipes [57]:

$$\phi_L = \left[1 + \frac{X}{C[F_d]^n}\right]\left(1 + \frac{12}{X} + \frac{1}{X^2}\right)^{1/2} \tag{22}$$

$$F_d = Fr\left(\frac{d}{D}\right)^{0.1} = \frac{U_L^2}{gd}\left(\frac{d}{D}\right)^{0.1} \quad \text{for } F_d \leq 0.3, \quad C = 7.79, \quad \text{and} \\ n = 0.576; \quad F_d < 0.3, \quad C = 13.56, \quad \text{and } n = 1.3 \tag{23}$$

where ϕ is the pressure drop multiplier, and X is the Lockhart–Martinelli parameter.

For vertical helicoidal pipes [58]:

$$\frac{\phi_L}{\left(1 + \frac{20}{X} + \frac{1}{X^2}\right)^{1/2}} = 1 + \frac{X}{65.45F_d^{0.6}} \quad \text{for } F_d \leq 0.1 \tag{24}$$

$$\frac{\phi_L}{\left(1 + \frac{20}{X} + \frac{1}{X^2}\right)^{1/2}} = 1 + \frac{X}{434.8F_d^{1.7}} \quad \text{for } F_d > 0.1 \tag{25}$$

where F_d is defined as

$$F_d = Fr\left(\frac{d}{D}\right)^{1/2}(1 + \tan\beta)^{0.2} = \frac{U_L^2}{gd}\left(\frac{d}{D}\right)^{1/2}(1 + \tan\beta)^{0.2} \quad \text{for} \\ d = 19.1\ \text{mm}, \quad D = 340\ \text{mm}, \quad \text{and } b = 0.5^\circ \tag{26}$$

The effects of coil geometries and the flow rates of air and water on two-phase flow pressure drop in annular vertical and horizontal helical pipes with three different diameters of inner and outer tubes were investigated by Xin et al. [49]. The experiments were performed for superficial water and air Reynolds numbers in the range of 210–23,000, 30–30,000, respectively. Correlations of the pressure drop multiplier for two-phase flow in horizontal and vertical annular helicoidal pipes were developed from the experimental data as follows:

For horizontal helical coil tube:

$$\phi_L^2 = \left(1 + \frac{10.646}{X} + \frac{1}{X^2}\right) \tag{27}$$

For vertical helical coil tube:

$$\phi_L = \left(1 + \frac{0.0435X^{1.5}}{F}\right)\left(1 + \frac{10.646}{X} + \frac{1}{X^2}\right)^{1/2} \tag{28}$$

where $F = Fr^{0.9106}e^{0.0458(\ln Fr)^2}$, $1.61 < d_{o,in}/d_{i,out} < 1.67$, $21 < D/(d_{o,in} - d_{i,out}) < 23$, Fr is the Froude number, defined as:

$$Fr = \frac{U_L^2}{g(d_{o,in} - d_{i,out})} \tag{29}$$

An experimental study on three-phase flow was carried out by Chen and Guo [59]. The flow patterns and pressure drops of oil–air–water three-phase flow in helically coiled plexiglass tubes with two different coil diameters were studied. The effects of flow rates and liquid properties on the pressure drop were investigated. The results showed that flow characteristics can be classified into more than four flow patterns and some flow regime maps were presented. Based on the experimental data, correlations of pressure drop were also proposed. Kang et al. [29] discussed the effects of cooling wall temperature on the condensation pressure drop characteristics of refrigerant HFC-134a in annular helical coil tubes. The results showed that the pressure drop increased slowly with increasing mass flux. The measured data were compared with those of Eckels and Pate's correlation. The correlations of pressure drop in terms of the refrigerant mass flux were also proposed.

Ju et al. [50] determined the two-phase flow pressure drop in small bending radius helical coil-pipe used in an HTR-10 steam generator. Based on the uniform flow formula with a correction factor, a formula for frictional pressure drop was proposed as:

$$\Delta P = f\frac{L}{d}\frac{\rho\omega_o^2}{2}\left[1 + x\left(\frac{\rho_G}{\rho_L - 1}\right)\right]\Psi \tag{30}$$

where f is the friction factor, ω_o the flow velocity, x the average steam content, Ψ the unevenness correction factor

$$\Psi = \left(1.29 + A_n x^n\right)\left\{1 + x\left[\left(\frac{\mu_L}{\mu_G}\right)^{0.25} - 1\right]\right\} \quad \text{for } 2.5 < P < 4.5 \text{ Mpa}, \quad 8 < D/d < 9.3, \tag{31}$$

where $A_1 = 2.19$, $A_2 = -3.61$, $A_3 = 7.35$, and $A_4 = -5.93$

Guo et al. [51] studied the pressure drops of steam–water two-phase flows in two helical coiled tubes with four different helix axial inclinations. The results showed that the system pressure and mass quality had significant effect on the two-phase pressure drop. A correlation based on the correlation of Chen for boiling two-phase flow frictional pressure

drop in helical coiled tubes was obtained

$$\phi_{\mathrm{Lo}}^{2}=\frac{\Delta P_{\mathrm{tp}}}{\Delta P_{\mathrm{o}}}=\psi\psi_{1}\left(1+x\left[\left(\frac{\rho_{\mathrm{L}}}{\rho_{\mathrm{G}}}-1\right)\right]\right) \tag{32}$$

$$\psi=1+\left[\frac{\left(x(1-x)\left(\frac{100}{G}-1\right)\frac{\rho_{\mathrm{L}}}{\rho_{\mathrm{G}}}\right)}{\left(1+(1-x)\left(\frac{\rho_{\mathrm{L}}}{\rho_{\mathrm{G}}}-1\right)\right)}\right] \tag{33}$$

$$\psi_{1}=142.2\left(\frac{P}{P_{\mathrm{cri}}}\right)^{0.62}\left(\frac{d}{D}\right)^{1.04} \tag{34}$$

Compared with the numerous investigations of single-phase flow, gas–liquid two-phase flow and oil–gas–water three-phase flow through the helical coil tubes, only limited information regarding the gas-solid two-phase flow is available in literature. Weinberger and Shu [60] determined the pressure drop of gas-solids flow in helical coil with two different helix diameters. The measured data were compared with results predicted by the 90° bend correlation. The results showed that the variations of solid pressure drop depend on solids flow rate, helix radius, and loading ratios. In their second paper [61], they determined the transition velocities as a function of bend or helix radius and solids flow rate. The measured velocities were compared with those predicted from modified horizontal flow correlation. It showed that the transition velocities decreased with increasing bend radius and solid flow rate. The modified correlation was shown to fit the measured data.

3. Spirally coiled tube

3.1. Heat transfer characteristics

3.1.1. Single-phase flow

The single-phase heat transfer characteristics in spirally coiled heat exchangers have rarely been investigated. Ho et al. and Wijeysundera et al. [62–65] used the relevant correlations of the tube-side and air-side heat transfer coefficients reported in literature in the simulation to determine the thermal performance of the spiral-coil heat exchanger under cooling and dehumidifying conditions. Experiments were performed to verify the simulation results.

Recently, an average in-tube heat transfer coefficient in a spirally coiled tube was proposed by Naphon and Wongwises [66]. The test section was a spirally coiled heat exchanger consisting of six layers of concentric spirally coiled tubes. The experiments were performed under cooling and dehumidifying conditions and considered the effects of inlet conditions of both working fluids on the in-tube heat transfer coefficient. The results obtained from experiments were compared with those calculated from other correlations. A new correlation for the in-tube heat transfer coefficient for spirally coiled tube was

proposed as follows:

$$Nu = 27.358De^{0.287}Pr^{-0.949} \quad \text{for } 300 \le De \le 2200,\ Pr \ge 5 \tag{35}$$

In their second and third papers, Naphon and Wongwises [67,68] developed a mathematical model to determine the performance and heat transfer characteristics of spirally coiled finned tube heat exchangers under wet-surface conditions. In addition, the heat transfer characteristics and performance of a spirally coiled heat exchanger under dry-surface conditions were studied theoretically and experimentally. The calculated and measured results were in reasonable agreement.

4. Other curved tubes

4.1. Heat transfer characteristics

4.1.1. Single-phase flow

Kalb and Seader [69] solved the full continuity and Navier–Stokes equations to analyze the effect of the curvature ratio (radius of bend/inside radius of tube) on fully developed heat transfer in curved circular tubes with a uniform-wall-temperature. They proposed a correlation for the fully developed Nusselt number as follows:

$$Nu = 0.836De^{0.5}Pr^{0.1} \quad \text{for } De \ge 80, \quad 0.7 < Pr < 5 \tag{36}$$

The influences of centrifugal and buoyancy forces on the fully developed laminar flow in horizontal and vertical curved pipes under constant temperature gradient in the direction of the axis were studied by Yao and Berger [70]. At sufficiently far distances from the pipe entrance, to avoid inlet-length effects, the flow pattern, local shear stress distribution and heat transfer mechanism were presented. Prusa and Yao [71] considered the combined effects of both buoyancy and centrifugal forces on the flow field and temperature distribution in a hydrodynamically and thermally fully developed flow for horizontal heated curved tubes. The numerical results indicated that the mass flow rate was drastically reduced because of the secondary flow. Higher curved tube and overheating conditions lead to a decrease of the total heat transfer rate. A flow-regime map comprised of the dominant centrifugal force region, the dominant buoyancy and centrifugal forces region, and the dominant buoyancy force region, was presented. Lee et al. [72] presented the influence of buoyancy on steady fully developed laminar flow in curved tubes with an axially uniform heat flux and peripherally uniform wall temperature covering a wide range of Prandtl, Dean and Grashof numbers and curvature ratios. The results indicated that the average Nusselt number, the local Nusselt number distribution around the periphery, and the orientations of the secondary flow were affected by the buoyancy force.

Goering and Humphrey [73] solved the fully elliptic Navier–Stokes and energy equations to analyze the effects of buoyancy and curvature on the fully developed laminar flow through a heated horizontal curved. Buoyancy and curvature effects and thermal boundary conditions were studied. The test sections with constant peripheral tube-wall temperature and constant peripheral heat flux were examined. Flow velocity

and temperature contours were presented. Li et al. [74,75] applied the renormalization group (RNG) $k-\varepsilon$ model for considering the three-dimensional turbulent mixed convective heat transfer in the entrance region of a curved pipe [74]. The relative magnitude of buoyancy and centrifugal effects on the secondary flow was characterized by using a new proposed dimensionless parameter. Comparison between the results obtained from the model and experiments showed good agreement. In addition, they studied numerically the developing turbulent flow and heat transfer characteristics of water near the critical point [75]. Based on the constant wall temperature with and without buoyancy effect, the velocity, temperature, heat transfer coefficient, friction factor distribution, and effective viscosity were presented and discussed.

A mathematical model based on the equations of conservation of mass, momentum and energy was determined by Targett et al. [76]. They studied the fully developed angular flow and fully developed convection in the annulus between two concentric cylinders. The results determined by using a finite-element representation and the FIDAP^{TM} code, showed that the heat flux density ratios as well as the Dean number are dependent on the Nusselt number. Wang and Cheng [77] studied numerically the combined free and forced convective heat transfer in a rotating curved circular tube with uniform wall heat flux and peripherally uniform wall temperature. The effects of curvature, rotation and heating/cooling on the temperature distribution, and Nusselt number were presented under steady, hydrodynamically, and thermally fully developed laminar flows.

Yang et al. [78] studied the effects of the flow rate, the Prandtl number, the pipe-period and the pipe-amplitude on the heat transfer for a laminar flow in a pipe with periodically varying finite curvature. The results showed that enhancement of the heat transfer rate could be achieved by increasing the amplitude and/or a decrease in the wavelength of a periodic wavy pipe. Nigam et al. [79] solved the governing equations for fully developed laminar flow and heat transfer of Newtonian and power law fluids in the thermal entrance region of curved tubes. The secondary velocity profile, temperature profile, Prandtl number, and power law index were provided. Results for friction factors, asymptotic Nusselt numbers and Nusselt numbers in the thermal entrance region were computed. Satisfactory agreement was obtained between the experimental data and numerical results. Andrade and Zaparoli [80] employed the finite element method for solving the mass, momentum and energy equations to investigate the fully developed laminar flow of the heating and cooling of water in a curved duct with temperature-dependent viscosity. The results showed that when the fluid was cooled with variable viscosity assumption, the Nusselt numbers were lower than those of the constant properties. This might be the decrease of the secondary flow effect due to the higher viscosity values.

The turbulence and heat transfer in two types of square sectioned U-bend duct flows with mild and strong curvature, using recent second moment closures, were predicted by Suga [81]. A two-component limit turbulence model and the wall reflection free model were presented. The results showed that the two-component limit turbulence model was reliable in the case of strong curvature.

Yang and Chiang [82] studied the effects of the Dean number, Prandtl number, Reynolds number and the curvature ratio on the heat transfer for periodically varying-curvature curved-pipe inside a larger diameter straight pipe to form a double-pipe heat exchanger. The results showed that the heat transfer rate increased by up to 100% as

compared with a straight pipe. All of the experimental data were regressed to obtain the following correlation of the Nusselt number.

For laminar flow ($Re < 2000$):

$$Nu = 0.185 De^{0.325} \delta^{-0.157} Pr^{0.234} \quad \text{for } 2.5 \times 10^4 \leq De \leq 6 \times 10^5, \quad 0.050 < \delta < 0.096, \quad 3.9 < Pr < 4.5 \tag{37}$$

For turbulent flow ($Re > 2000$):

$$Nu = 2.87 De^{0.4} \delta^{-0.203} Pr^{0.386} \quad \text{for } 2.1 \times 10^6 \leq De \leq 5.5 \times 10^7, \quad 0.050 < \delta < 0.096, \quad 4.0 < Pr < 5.2 \tag{38}$$

4.2. Flow characteristics

4.2.1. Single-phase flow

The velocity profiles for the laminar flow of a Newtonian liquid in curved tubes were determined by Soeberg [83]. A technique based on the symmetry of the secondary-flow field was used for solving the equations of fluid motion and continuity of a fully developed, steady, isothermal and incompressible fluid. The results revealed that for $De < 16$, the secondary flow influenced the shape of the harmonics of the axial velocity. For $De > 16$, the harmonics changed shape and amplitude. The velocity profile at the center became flatter as the Dean number increased. For $De > 100$, the Coriolis force influenced the stability of the laminar-flow field, moving the transition point to turbulent flow. Yanase et al. [84] analyzed the stability of two-vortex and four-vortex solutions of flow through a slightly curved circular tube by using the Fourier–Chebyshev spectral method for Dean numbers ranging between 96 and 10,000. The results showed that the two-vortex solution was stable in response to any small disturbances, while the four-vortex solution was unstable to asymmetric disturbances.

Goering and Humphrey [73] studied the effect of curvature and buoyancy on flow characteristics and pressure drop of fully developed laminar flow through a heated horizontal curved tube with a constant peripheral tube wall-temperature and constant peripheral heat flux using the full-elliptic Navier–Stokes and energy equations. Zhang et al. [85] studied the combined effect of the Coriolis and centrifugal forces on the flows in rotating curved rectangular ducts. The effects of the force ratio and the aspect ratio of the cross-section on the characteristics of the secondary flow, the axial flow and the friction factor were considered. Recently, Yanase et al. [86] used a spectral method to analyze the laminar flow in a curved rectangular duct over a wide range of the aspect ratio using the Newton–Raphson method. Five branches of the steady solutions were formed and linear stability characteristics were studied for all steady solutions.

Jain and Jayaraman [87] studied the effects of constriction combined with constant curvature of the center line on fully developed steady flow of a fluid through a curved tube. The phenomenon of secondary flow, shear stress and the increased impedance due to constriction were presented. Rodman and Trenc [88] investigated the influence of

the channel curvature on the pressure drop in laminar oil-flow in curved rectangular channel-coils with different geometrical aspect ratios and different curvatures. The results obtained from this work were compared with experimental work of Baylis, Ludwieg, Cheng et al., and Cheng and Akiyama [89–92]. The regression curve for the pressure drop was obtained as follows:

$$f_c \cdot Re = 2.4629De^{1/2}(1 - 18.553De^{-1/2}) + 275.38De^{-1} - 1015.9De^{-3/2} \quad \text{for } 100 < De < 800,\ 1 < H/w < 5,\ 7 < R/D_h < 15 \tag{39}$$

The product of $f \cdot Re$ was calculated by the equation

$$f \cdot Re = \frac{\Delta P \cdot D_h^2}{2 \cdot l \cdot \rho \cdot v \cdot U_c} \tag{40}$$

where H is the height of the channel, w is the width of the channel, R is the radius of the curvature, U_c is the mean axial flow velocity, D_h is the hydraulic diameter of the channel.

Yang and Chiang [82] studied the pressure drop of water flowing through a varying-curvature curved-pipe inside a large diameter straight pipe to form a double-pipe heat exchanger. The effects of the Dean number, Prandtl number, Reynolds number and the curvature ratio (δ) on the friction factors were discussed. As compared with a straight pipe, the results indicated that the friction factor increased by less than 40%. Based on the experimental data, correlations of the friction factor were proposed as follows:

For laminar flow ($Re < 2000$):

$$f_c = 739De^{-0.507}\delta^{0.988} \quad \text{for } 2.5 \times 10^4 \le De \le 6 \times 10^5, \quad 0.050 < \delta < 0.096, \quad 3.9 < Pr < 4.5 \tag{41}$$

For turbulent flow ($Re > 2000$):

$$f_c = 1.69De^{-0.159}\delta^{0.488} \quad \text{for } 2.1 \times 10^6 \le De \le 5.5 \times 10^7, \quad 0.050 < \delta < 0.096, \quad 4.0 < Pr < 5.2 \tag{42}$$

4.2.2. Two-phase flow

The only work concerned with two-phase flow in curved tubes is that of Gao et al. [93]. They simulated solid–liquid two-phase flows in two-dimensional curved channels. Effects of different particle size, liquid flow rate and coil curvature on the phase distribution characteristics were considered. Based on the numerical results, the dynamic effects and contributions to the phase separation of particle-subjected forces were presented, including centrifugal force, drag force, pressure gradient force, gravity force, buoyancy force, virtual mass force and lift force. In addition, the effects of secondary flow on the concentration distribution characteristics and phase separation of two-phase flow in helically coiled tube were experimentally investigated.

Table 1
Available in-tube heat transfer correlations

Authors	Conditions	Working fluids
Seban and Mclaughlin [94] cited by Guo et al. [26]	Helical coil $6000 < Re < 60{,}000$	Water
Roger and Mayhew [95], cited by Ho et al. [62]	Helical coil, turbulent flow	Water
Mori and Nakayama [96], cited by Ho et al. [62]	Helical coil, laminar flow, turbulent flow	Air
Dravid et al. [1]	Helical coil $5 < De < 2000$ $5 < Pr < 175$	Water, *n*-amyl acetate *n*-butanol, thylene glycol *n*-amyl alcohol
Kalb et al. [69]	Curved tube $De > 80$ $0.7 < Pr < 5$	Newtonian fluids
Oliver and Ashar [97] cited by Nigam et al. [79]	Helical coil $4 < De < 60$ $60 < De < 2000$	Newtonian, viscoelastic liquids
Havas et al. [22]	Helical coil $3.2 \times 10^3 < Re_o < 3.5 \times 10^5$ $1.3 \times 10^3 < Re < 1.6 \times 10^5$ $2.7 < Pr < 124$ $0.25 < d_a/D_v < 0.4$, $0.03 < d_o/D_v < 0.051$	Water
Mikaila et al. [98] cited by Xin et al. [49]	Helical coil, turbulent flow	–
Cengiz et al. [23]	Rotating helical coil	Air
Cengiz et al. [24]	Helical coil with spring inside $1265 < De < 2850$, $Pr = 0.7$ $1315 < De < 3200$, $Pr = 0.7$	Air
Bolinder and Sunden [48]	Helical square duct $De < 510$ $0.005 < Pr < 500$	–
Xin et al. [49]	Helical coil $20 < De < 2000$, $0.7 < Pr < 175$ $5 \times 10^3 < De < 10^5$, $0.7 < Pr < 5$ $0.0267 < d/D < 0.0884$	Air, water, ethylene-glycol
Guo et al. [26]	Helical coil, rotating helical coil $6 \times 10^3 < Re < 1.8 \times 10^5$ $2.5 \times 10^4 < Re < 1.25 \times 10^5$, $0.003 < f < 0.05$	Water
Kang et al. [29]	Helical coil $1100 < Re^* < 2500$	R-134a
Acharya et al. [6]	Coiled tube $50 < Re < 1000$ $0.1 < Pr < 10$	–
Nigam et al. [79]	Curved tube $2 < De < 830$, $30 < Pr < 450{,}000$	Non-Newtonian fluid
Yang and Chiang [82]	Curved pipe	Water

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Table 1 (*continued*)

Authors	Conditions	Working fluids
	$2.5\times10^4 \le De \le 6\times10^5$, $3.9 < Pr < 4.5$ $2.1\times10^6 \le De \le 5.5\times10^7$, $4.0 < Pr < 5.2$	
Lemenand and Peerhossaini [13]	Coiled tube $100 < Re < 300$ $30 < Pr < 100$	-

Table 2
Available pressure drop correlations for single-phase flow

Authors	Conditions	Working fluids
Ito [99] cited by Ali [52]	Curved pipe, laminar flow $13.5 < De < 2000$	Air, water
Ito [99] cited by Ali [52]	Curved pipe, turbulent flow, $Re(d/D)^2 > 6$	Air, water
Mori and Nakayama [35] cited by Ali [52]	Helical coil, laminar flow, $13.5 < De < 2000$	Air
Mori and Nakayama [96] cited by Ali [52]	Helical coil, turbulent flow	Air
Schmidt [100] cited by Ali [52]	Curved tube, laminar flow	–
Srinivasan et al. [101] cited by Ali [52]	Helical coil, $0.0097 < d/D < 0.135$	–
	$Re/\sqrt{(d/D)} < 30$ $30 < Re/\sqrt{(d/D)} < 300$ $30 < Re\sqrt{(d/D)} < Re_{cri}\sqrt{(d/D)}$ $Re > Re_{cri}$	
Tarbell and Samuels [9]	Helical coil, $20 < De < 500$, $2 < D/d < 30$	–
Ramana Rao and Sadasivudu [102] cited by Ali [52]	Helical coil, $0.0159 < d/D < 0.0556$, $Re < 1200$ $0.0159 < d/D < 0.0556$, $1200 < Re < Re_{cri}$ $0.0159 < d/D < 0.0556$, $Re_{cri} < Re < 27{,}000$ $0.0159 < d/D < 0.0556$	–
Mishra and Gupta [45,46] cited by Grundmann [44]	Helical coil, laminar, $1 < De < 3000$	–
	Helical, turbulent, $4500 < Re < 10^5$, $6.7 < D/d < 346$ $0 < p/D < 25.4$	
Manlapaz and Churchill [103] cited by Awwad et al. [57]	Helical coil	–
Hart et al. [47]	Helical coil, laminar, $0 \le Re \le Re_{cri}$	Air
Yanase et al. [84]	Curved tube, laminar	–
Liu and Masliyah [104] cited by Ali [52]	Helical coil, developing laminar	–
Ruffle (1994) (see Czop et al. [105]) cited by Guo et al. [51]	Turbulent flow	–
Xin et al. (1997)	Annular helicoidal pipe $35 < De < 20{,}000$	Air, water

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Table 2 (*continued*)

Authors	Conditions	Working fluids
	$1.16<d_{i,out}/d_{o,in}<1.67$ $21<D/(d_{i,out}-d_{o,in})<32$	
Ju et al. [50]	Laminar, $De<11.6$ Laminar, $De>11.6$, $Re<Re_{cri}$ Turbulent, $De>11.6$, $Re>Re_{cri}$	Water
Guo et al. [51]	$1.5\times10^5<Re<4\times10^5$, $300<G<4300$, G=mass flux, $D=132$, 256 mm, $d_i=10$, 11 mm, length=4836, 7560 mm	Water
Ali [52]	Helical coil $0.027<d/D<0.052$, $0.0445<p/D<0.43$	Water
Rodman and Trenc [88]	Curved tube, laminar $100<De<800$	Oil
Yang and Chiang [82]	Curved tube $2.5\times10^4\leq De\leq6\times10^5$, $3.9<Pr<4.5$ $2.1\times10^6\leq De\leq5.5\times10^7$, $4.0<Pr<5.2$	Water

Table 3
Available pressure drop correlations for two-phase flow

Authors	Conditions	Working fluids
Akagawa et al. [106] cited by Guo et al. [51]	Helical coil $d/D=0.0188$	Steam–water, chemical compound–water
Stepanek and Kasuri [55]	Helical coil	Air–water
Unal et al. [107] cited by Guo et al. [51]	Helical coil $0.00537<d/D<0.0217$	Steam–water
Chen and Zhou [108] cited by Guo et al. [51]	Helical coil $0.0198<d/D<0.076$	Steam–water
Nariai et al. [109] cited by Guo et al. [51]	Helical coil $d/D=0.024$	Steam–water
Rangacharyulu and Davies [56]	Helical coils, $0.0427<R_i/R_c<0.0541$	Air–water, air–glycerol, air–isobutyl alcohol
Hart et al. [47]	Helical coil, $d/D=0.0348$ $10<U_G<40$ m/s, $8\times10^{-4}<U_L<3\times10^{-2}$ m/s	Air–water, air–water–glycol
Award et al. [57]	Helical coil, $12.7<d_i<38.1$ mm, $330<D<670$ mm, $0.008<U_L<2.2$ m/s, $0.2<U_G<50$ m/s	Air–water
Xin et al. [58]	Helical coil, $12.7<d_i<38.1$ mm, $D=305$, 609 mm $0.008<U_L<2.2$ m/s, $0.2<U_G<50$ m/s	Air–water
Xin et al. [49]	Annular helical coil, $1.61<d_{i,out}/d_{o,in}<1.67$, $21<D/(d_{i,out}-d_{o,in})<32$ $210<Re_L<23{,}000$	Air–water

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Table 3 (*continued*)

Authors	Conditions	Working fluids
Chen and Guo [59]	Helical coil $d_i = 39$ mm $D = 265, 522.5$ mm	Oil–air–water
Kang et al. [29]	Helical coil, $100 < G_f < 400$ kg/m^2 s	R134a
Ju et al. [50]	Helical coil, $8 < D/d < 9.3$, $2.5 < P < 4$. 5 MPa, $200 < G < 1500$ kg/m^2 s	Steam–water
Guo et al. [51]	Helical coil, $1.5 \times 10^5 < Re < 4 \times 10^5$ $150 < G < 1760$ kg/m^2 s, $D = 132, 256$ mm, $d_i = 10, 11$	Steam–water

5. Conclusions

In this present study, curved tubes can be divided into three groups according to configurations of the tube curvature. The conclusions can be summarized as follow:

- For helical coil tubes, the above survey indicates that numerous theoretical and experimental works have been reported on single-phase heat transfer characteristics, single-phase and two-phase flow characteristics. Two-phase heat transfer characteristics have rarely been reported.
- For spiral coil tubes, although a few papers had been published, only one of these papers presented the correlation of the in-tube heat transfer coefficient. In addition, none of the papers presented the flow characteristics and pressure drop.
- For other curved tubes, single-phase, two-phase heat transfer characteristics and single-phase heat transfer characteristics have been numerously presented. But only one work reported on two-phase flow characteristics.

In addition, only some papers have presented the effects of the combined active and passive method on the enhancement of heat transfer rate and pressure drop. The study points out that although numerous studies have been conducted on the characteristics of heat transfer and flow in curved tubes, study on some types of curved tubes is limited, especially on spirally coiled tubes (Tables 1–3).

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